Combustion Acoustics Model Inside a Pratt & Whitney Lean Premixed Combustor

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Executive Summary

My UTSR Fellowship was spent at Pratt & Whitney in East Hartford CT. I had the great fortune to work with extremely motivated and friendly engineers in the Combustor, Augmentor & Nozzle (CAN) Module Center. My lead mentor was Al Veninger who leads the combustor development team for Power Systems. My work required me to work closely with many of the engineers including Bill Proscia and Urmila Reddy on combustion instabilities with their lean premixed (Dry Low NOx) combustor.

While I had a sufficient amount of background in combustion, I had not dealt with the specific aspects of combustion instability and acoustics before my time at P&W. The assignment I was given was both challenging, and rewarding. I feel like I have taken away from this project a great deal of experience and knowledge.

Outside of my technical work, I had the opportunity to follow along to product team meetings that discussed both the technical and non technical side of engineering a complex product for sale. Additionally I was able to tour the facility and observe the manufacture and testing of many of the components in a gas turbine engines.

This report outlines some of the basics on combustion instabilities and how dynamics can be applied to it. Additionally the report summarizes the development of the code, some of the difficulties in modeling certain aspects along with comparisons of predictions to measured data from a sector rig test.

It is hoped that this report and code that it will allow the user to understand the algorithm and the results from the code so that this code can be used as a design tool for the design of a new combustion system.

Introduction

Combustion instabilities are significant problems to the reliability and endurance of gas turbine engines. Generally there exist two types of instabilities: static instability, where the flame within the combustor does not stay anchored to a bluff body and is blown out of the combustor; and dynamic instabilities which are a result of complex interactions between the fluid dynamics and heat release in the flame inside the combustion chamber. Dynamic instabilities are present in many combustion devices including early liquid and solid rockets, jet engine afterburners and lean premixed combustion chambers.

Gas turbine engines using lean premixed combustors have the potential to produce less NOx and CO than traditional diffusion type combustors, both with and without water. Hence, there has been a big push for Dry Low NOx (DLN) combustors where ever increasing emissions standards are present, despite their inherent instability problems.

The focus of this project was to develop a low order acoustic model of the whole combustor system in hopes of understanding the fundamental reasons for changes in frequency when various parameters (flame temperature, power setting, and hardware geometry) were adjusted for a new combustor design at Pratt and Whitney.

Background on Combustion Dynamics

Combustion dynamics is a challenging topic to understand and more so to model. Dynamics occur when a small perturbation in the combustor flow field causes another perturbation in the flame. Consider the diagram in Figure 1 when a momentary drop in fuel flow from the injector occurs. This may create an extra lean pocket of fuel/air which when burned causes a small fluctuation in the heat release within the flame. This heat release fluctuation will itself create a small perturbation in pressure (a sound wave) which then propagates back to the fuel injector and induces another perturbation in the fuel flow. This feedback process will then continue on thereafter.

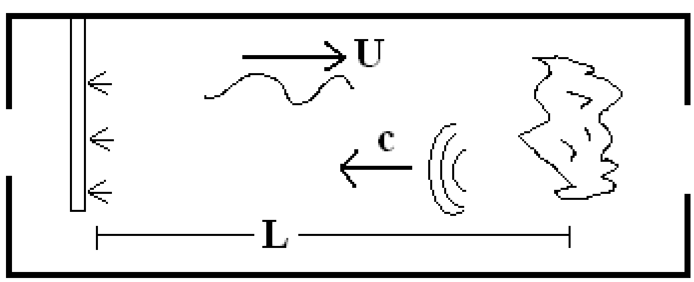


Figure 1: Illustration of Oscillation Feedback Loop

One way to model this is through a feedback loop as shown in Figure 2. Here the original input is u which could be either the flow velocity, fuel/air ratio or another parameter. The input u passes through the transfer function, G(s), which can be thought of the aerodynamics or chemistry in the flow field that creates the initial fluctuation q'. This perturbation is then fed back through H(s) which is the transfer function between heat release perturbation q' to the new pressure perturbation p' and finally on back to the original point and summed with the original input u.

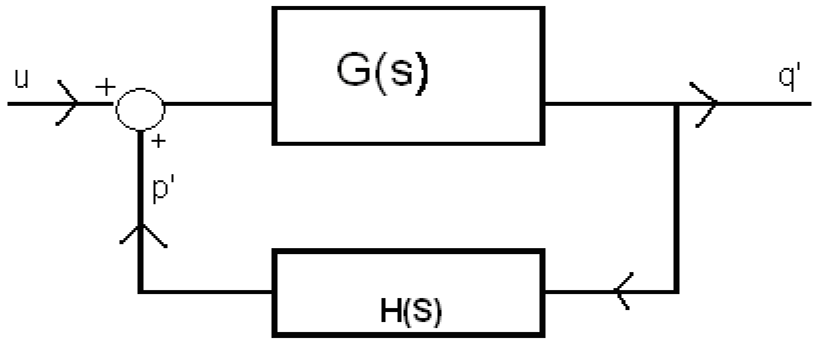


Figure 2: Control Loop for Combustion Instability Process

If this loop has negative feedback (positive damping), the pressure perturbation will eventually die out, p'=0. However if the system has positive feedback (the feedback results in driving the system to larger and larger fluctuations) the system will eventually go unstable.

In the engine, these instabilities can result in pressure fluctuations up at high frequency (100 Hz and up) which induce vibrations, noise and eventually engine damage. Hence it is of critical importance to try and either eliminate the cause of the pressure perturbation, or if that is not possible, to provide sufficient damping to the system to mitigate perturbations to acceptable levels.

The frequencies or the period at which the oscillations will occur are dependent upon the geometry and flow conditions within the combustor. This can be illustrated based again on Figure 1. For instance, the convective time for the fuel/air pocket to propagate downstream from the injector to the flame is a function of the combustor length and flow rate:

$$\tau_1 = \frac{L}{U},$$

where U is the flow rate and L is the combustor length.

While the speed at which the sound waves propagate upstream is dependent on the temperature:

$$\tau_2 = \frac{L}{c} = \frac{L}{\sqrt{\gamma RT}},$$

where c is the sound speed which is equal to $\sqrt{\gamma RT}$ for an ideal gas where γ is the specific heat ratio, R is the gas constant an T is temperature.

The period of oscillation will be the time for sum of the two times $T = \tau_1 + \tau_2$ while the frequency is the inverse of the period, $f = \frac{1}{T}$. Therefore, varying any of these parameters (length, temperature, flow rate etc...) will affect the frequency of oscillations.

Two approaches to mitigating excessive pressure perturbation are to either design the combustor to eliminate the possibility of positive feedback; or accept some feedback but to provide sufficient damping to keep the pressure fluctuations to small values (less than 1 psi). In this project, the latter was attempted by sizing Helmholtz resonators to the frequency of oscillations to damp out the pressure fluctuations.

Development of Acoustic Model

A Matlab code was written during this project that would incorporate all the parts of the combustor system that would affect the frequency. It is the goal of this Matlab code to accurately predict the frequencies in the combustion chamber with various hardware geometry and operating conditions. The combustor model is shown schematically in Figure 3. The model incorporates the combustor volume, the volume of the Helmholtz resonators, the inlet and exit admittances, and also the volume of the diffuser case (not shown).

Because the frequency is fairly low for the bulk mode, the wavelengths of the pressure waves are sufficiently long that we can assume that the pressure within the volume at any instance is spatially uniform.

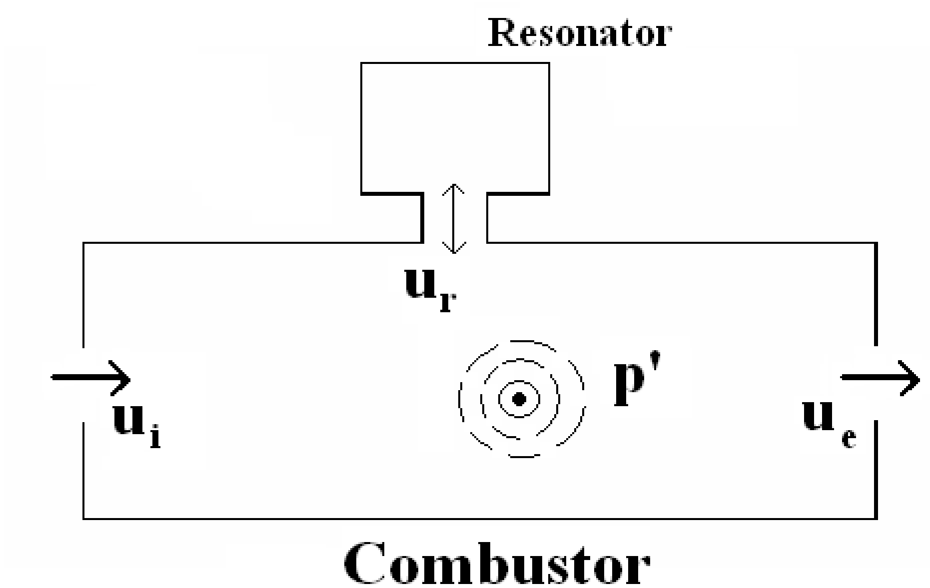


Figure 3: Bulk Mode Model for Combustor Volume

Linearized Equations

The pressure and heat release can be modeled in a simple linear manner through small perturbations to the mean pressure or heat release:

$$p = \overline{p} + p' \quad \text{and} \quad q = \overline{q} + q'$$

Where p is the total pressure, $\overline{p}$ is the mean pressure and p' is the pressure perturbation, and similarly for heat release, q.

Through this analysis a set of coupled linear ordinary differential equations can be set up that describes the relation between the perturbation in pressure, inlet and exit admittances and resonators admittances.

For example, the pressure perturbation can be related to velocity perturbation through the equation:

$$p' = \rho c^2 u'$$

, (where ρ the density and c is is the sound speed.) The admittance is the ratio of u' to p' (u'/p') and the impedance is the inverse (p'/u').

Additionally, the relation between heat release and port admittances with pressure fluctuation was approximated by the formula:

$$\frac{d\left(p'\right)}{dt} \sim \frac{\delta q}{\delta p} p' \approx N \frac{\overline{q}}{\overline{p}} p',$$

where γ is the ratio specific heats, R is the ideal gas constant, V is volume, Cp is specific heat, g is gravity constant, and N is a combustion gain, or coefficient that is adjusted to match data. The gain could be thought of as the 'intensity' of the heat release to pressure perturbation. A system that is more stable will be able to tolerate a higher intensity (or gain) than a marginally stable system. This model links disturbances in the pressure field to heat release fluctuations.

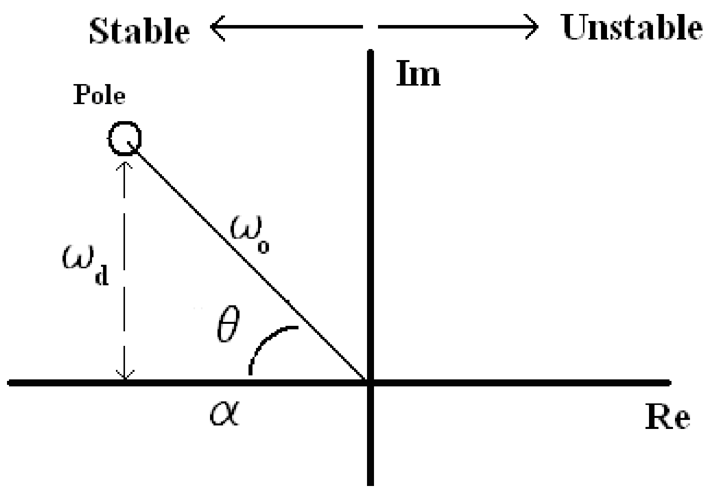
State Space

The system of equations can then be placed into a state space matrix form ($\dot{\mathbf{X}} = \mathrm{A}\mathbf{X}$) as shown in Figure 4. The matrix shown is a simplified (a 3x3) of the full system which includes other variables for heat release, diffuser, and resonators etc.

$$\frac{d}{dt}\begin{vmatrix} \rho c u_i \\ \rho c u_i \\ p' \end{vmatrix} = \begin{bmatrix} \frac{-M_i c_i}{l_i} & 0 & \frac{c_i}{l_i} \\ 0 & \frac{-M_e c_e}{l_e} & \frac{c_e}{l_e} \\ -\frac{A_e c_e}{V} & -\frac{A_e c_e}{V} & 0 \end{bmatrix} \cdot \begin{vmatrix} \rho c u_i \\ \rho c u_i \\ p' \end{vmatrix}$$

Figure 4: State Space form for simple model

The advantage of using state space analysis is that the eigenvalues, $\lambda = (A - sI)$, for the A matrix (the 3x3 in Figure 4), can be calculated simply in Matlab. There will be *n* eigenvalues will be for an *n* x *n* matrix. The eigenvalues are the roots of the denominator of the transfer function of the system, commonly called the poles.   These eigenvalues are generally a complex number, consisting of both real and imaginary parts. When the value is plotted on a Re-Im plot as in Figure 5, the components can be broken down into the damped natural frequency, ω_d (the imaginary part) and the damping ratio (real part), ζ (real part).



Re(eigenvalue) = α , Im(eigenvalue) = ω_d

$$\omega_d = \pm\omega_o\sqrt{\zeta^2 - 1} \qquad \textbf{damping ratio} = \zeta = \cos(\theta) = \frac{\alpha}{\omega_o}$$

Figure 5: Breakdown of Complex Pole to Frequency and Damping

Each eigenvalue will have its own unique position on the Re-Im plane and therefore have unique frequency and damping. As long as the damping ratio ζ is positive, (the pole remains on the left side of the Imaginary axis) this eigenvalue is stable. If the pole is on the right hand side, then it is unstable.

To understand why the damping must be positive for stability, consider the general form of the solution to these equations:

$$p' = A \cdot e^{-\alpha t} \cdot \sin(\omega_d t)$$

,where A is a general coefficient.   If the growth is positive, the exponential term will go to zero as time increases (meaning the perturbation is eliminated) while if the damping is negative the exponential term will increase toward infinity (perturbation are amplified) as time increases. In reality, the perturbation will never approach infinity as non-linear affects will limit the actual magnitude of the pressure perturbation, however these equations do a good job predicting if the system is stable (perturbation will not persist) or unstable (perturbation will exist with a finite amplitude). For this reason, only frequencies and damping ratios are predicted in this simple linear model and not amplitudes of pressure perturbations.

Resonators

As has been mentioned previously, system damping is required for stability. While perturbations may exist in the system, as long as the energy absorbed by the system is greater than the energy created in the heat perturbations the system can have finite but still acceptable small perturbations. One common method for adding damping to the system is through the use of a series of side branch Helmholtz resonators.

A resonator works by absorbing energy through fluctuations inside its neck. As seen in Figure 3, when the combustor pressure decrease density and fluctuations induces some of the air in the resonator to escape outside, however this creates a lower pressure zone within the resonator volume which forces air back in. The velocity within the neck will then oscillate back and forth (absorbing energy from the system) at a characteristic frequency given by:

Equation 1: Relation between resonator frequency and geometry

$$\omega_o = c\sqrt{\frac{A_N}{VL_N}}$$

, where A_N is the neck area, V_N is the volume, and L_{NT} is the effective neck length and c is the sound speed. The resonator works under the same principles as when one blows air past a 2-Litre bottle of soda and hears a low hum.

Varying Combustion Gain

The first tests performed varied the combustion gain in model without resonators to look at the combustor's inherent stability. It was found that the increase in combustion gain results in a decrease in damping (see Figure 6) and a decrease in frequency (see Figure 7). For this system a gain of N=4 results in the system becoming marginally stable. Any higher gain and the system will go unstable, and any less gain and the system will remain stable. Also at this threshold gain the normalized frequency of the system is 1.08. For an actual combustor the system damping ratio will lie somewhere in the slightly unstable region (0 to -0.15).

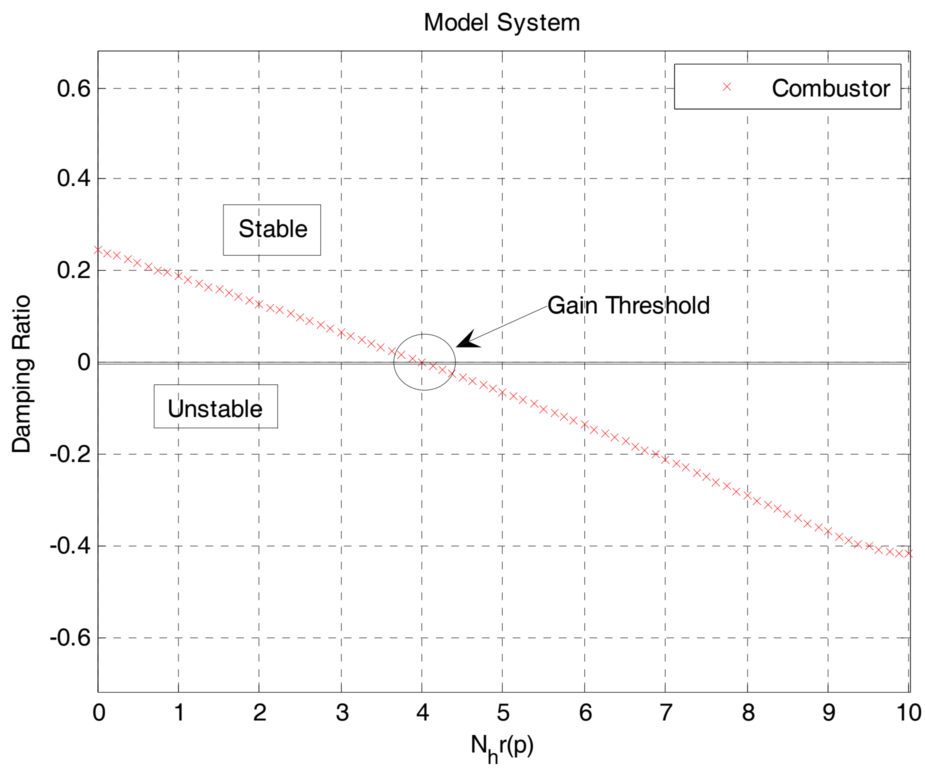


Figure 6: Damping ratio vs. combustion gain (without resonators)

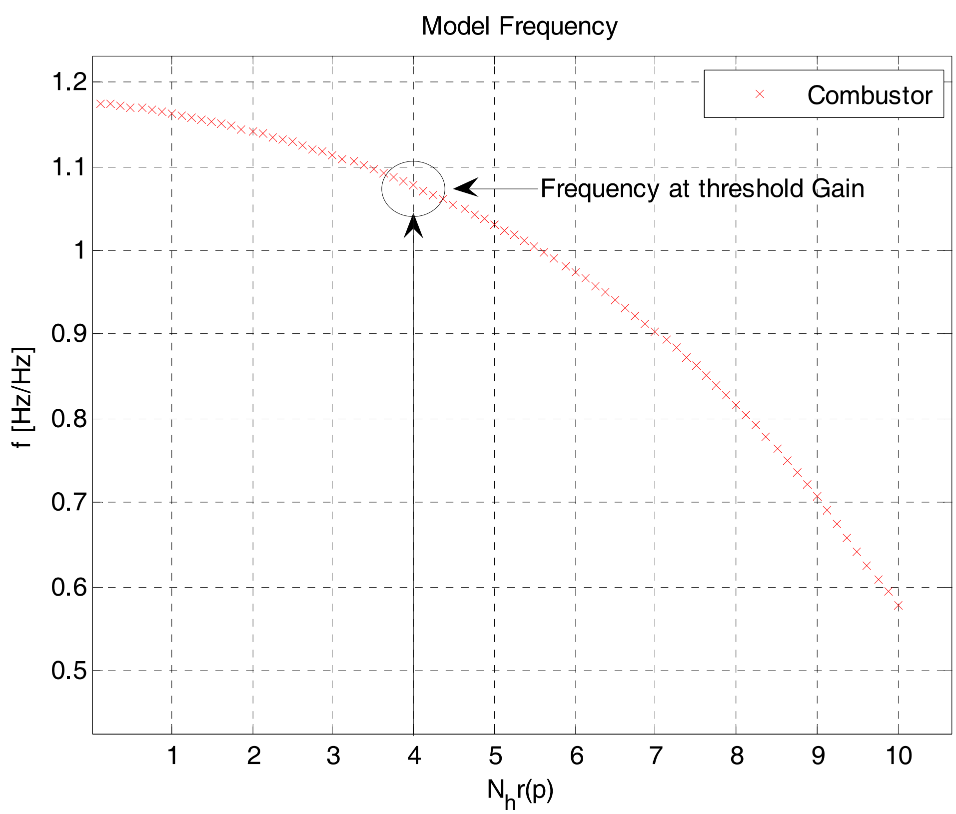


Figure 7: Normalized frequency vs. combustion gain (without resonators)

Tuning Resonators

In the next set of runs, the resonator volumes were varied in an isolated case to find the optimal volume. By increasing the resonator volume, the natural frequency will drop. Therefore one method to retune the resonators is to vary volume until its frequency matches of the frequency of combustion oscillation.

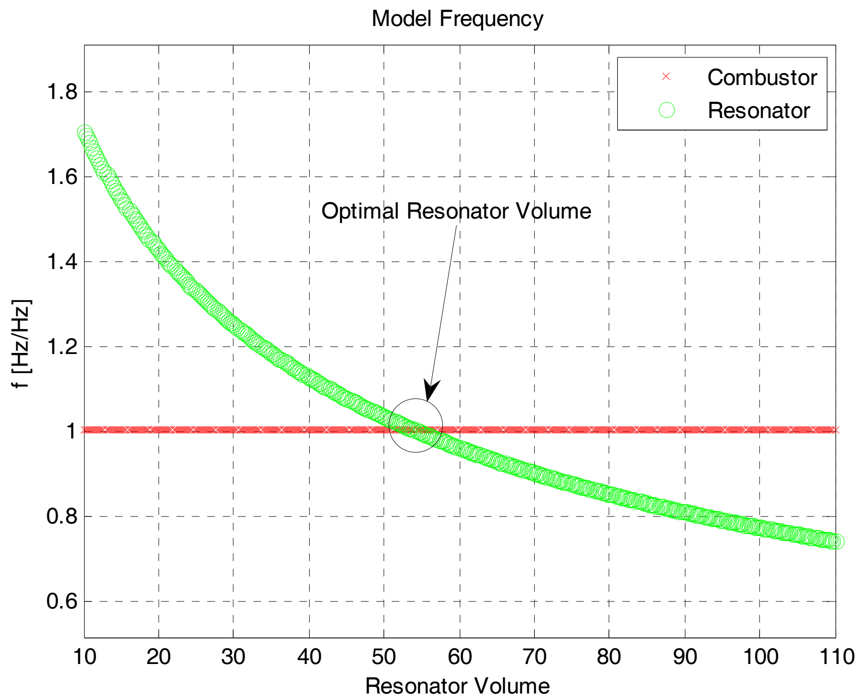


Figure 8: Frequency with varying resonator volume (isolated)

Figure 8 show normalized frequency of the resonator (green curve) as the volume is varied. Superimposed on these plots as well are the normalized frequency of engine combustor oscillation (red curve) at its nominal volume and heat release. According to the figure, the ideal volume for the resonator would be 55 in^3 as at this volume, the frequency of the resonator matches the frequency of combustion oscillation.

System with Damping

Now that the optimal damping volume has been determined, the run above with varying combustion gain was redone but with the resonators added into the system. The results are shown in Figures 10 and 11. Where as in the old system, at gain of N=4 the system was marginally stable (damping ratio = 0); now with the same gain the system has a damping of +0.08, giving some stability. Hence the resonators did their job and added enough damping to the system to keep it stable. A gain of 6 is now required to drive the system unstable. Also notice that the normalized frequency of combustion oscillation at the threshold gain changed from 1.08 (without resonators) to 1.25 when resonators were added to the system.

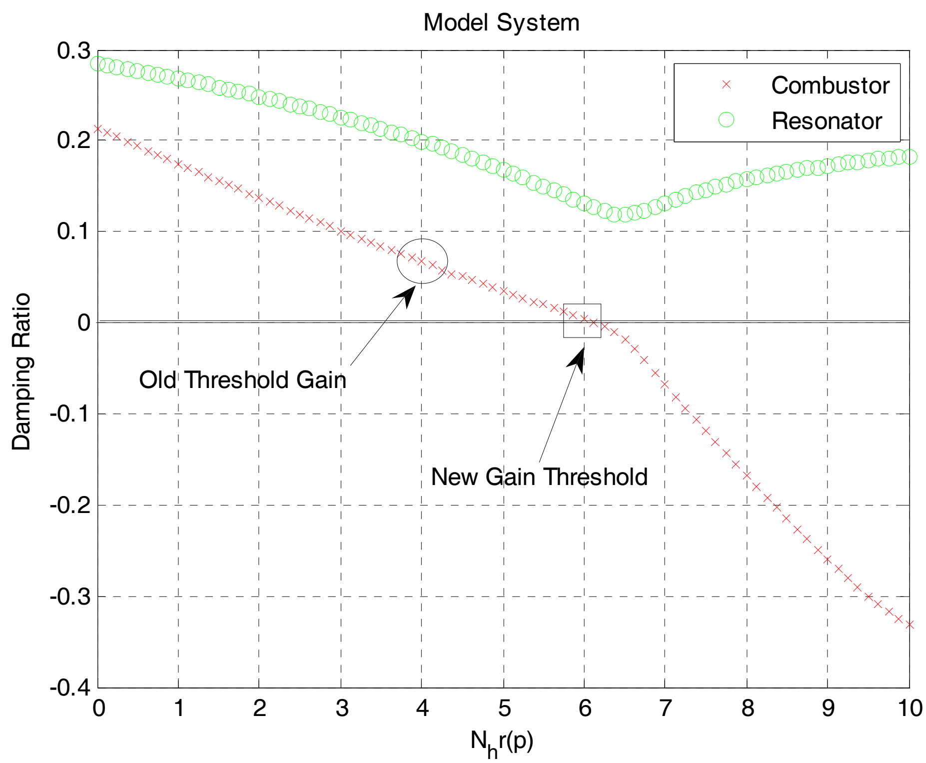


Figure 9: Damping ratio vs. combustion gain (with resonators)

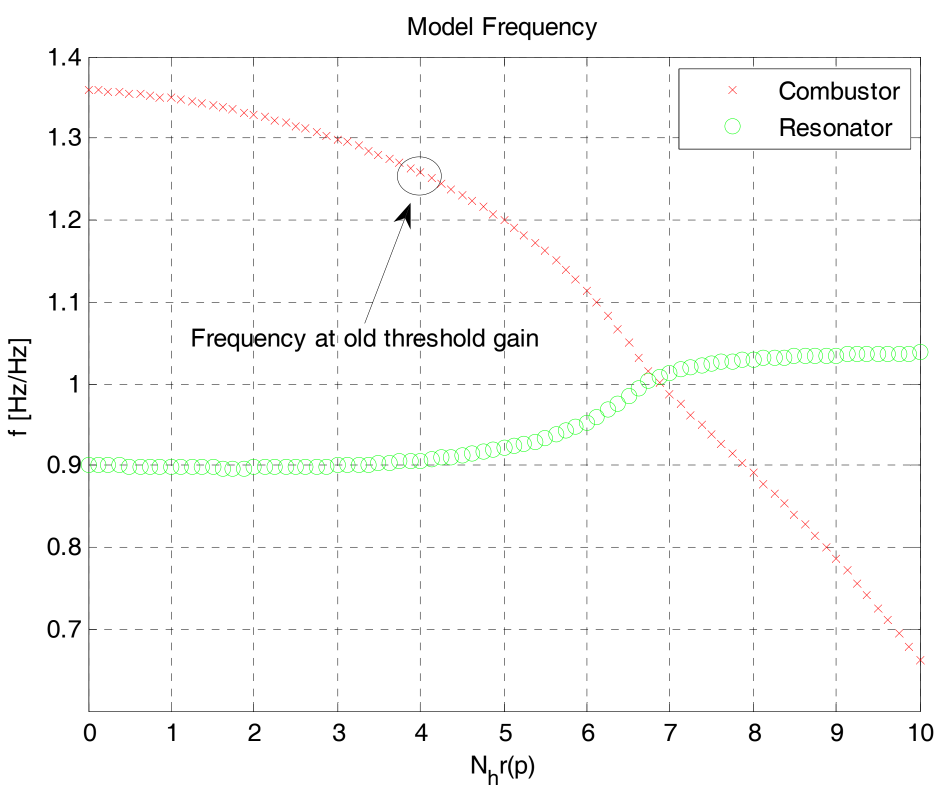


Figure 10: Normalized frequency vs. combustion gain (with resonators)

Optimizing Damping Ratio

A second parameter to optimize on the resonator is the damping. A too lightly damped resonator will not absorb enough energy from the system, and an overly damped resonator will make it difficult for air to pass through the neck, also hindering its ability to absorb energy.

The root locus in Figure 11 shows as the damping from the poles increases (moves to the left) which pulls the combustor poles left (but only so far before they double back). In Figure 12 which gives the combustor's damping ratio as the resonator damping ratio is varied, clearly shows a maximum damping for the combustor (red curve) around a resonator damping ratio of 0.28. Hence the optimal damping for this resonator is 0.28. Figure 13 also shows that the frequency of combustion oscillation will also vary for different resonator damping ratios.

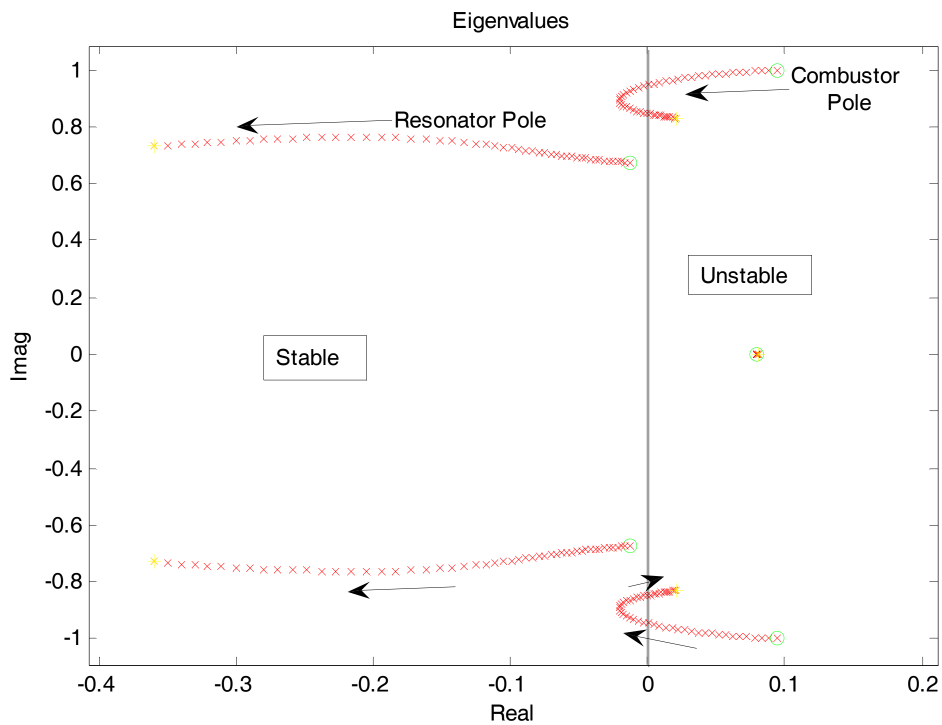


Figure 11: Root locus for varying resonator damping ratio.

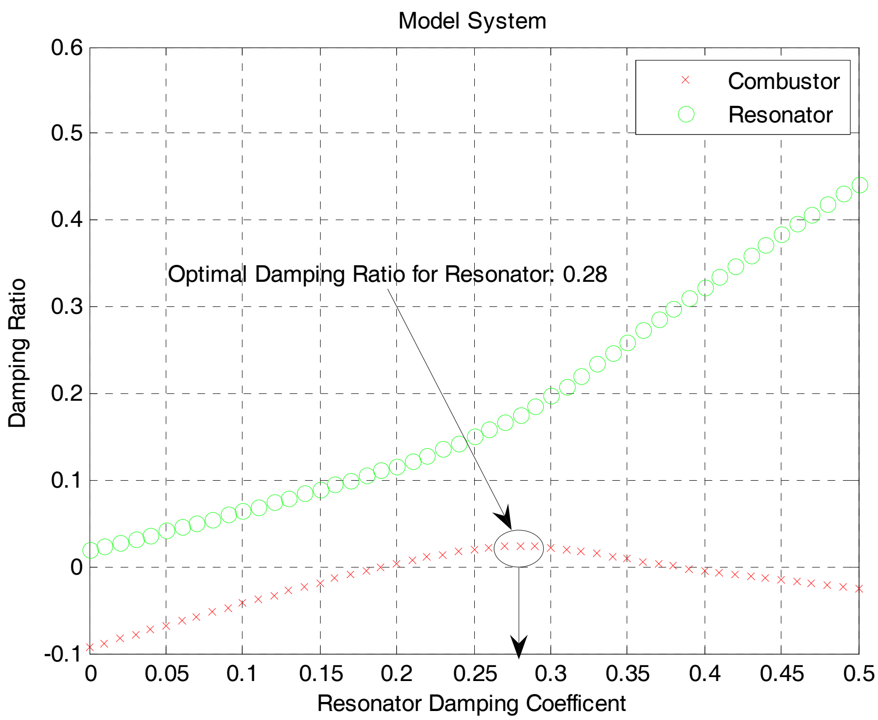


Figure 12: Damping ratio for varying resonator damping ratio

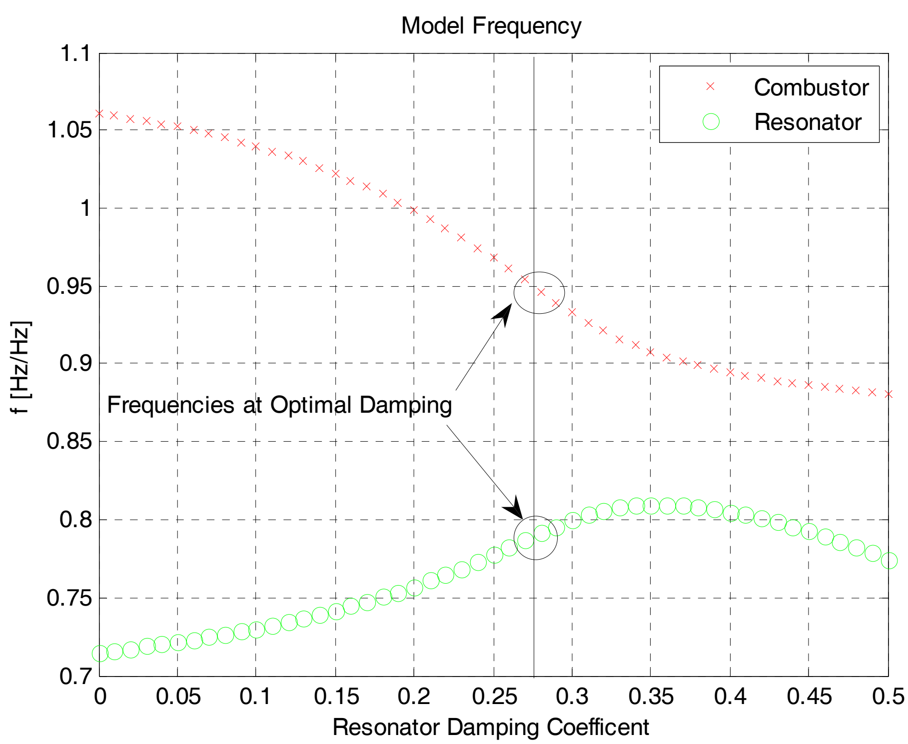


Figure 13: Frequency for varying damping ratio

Flame Temperature

The engine will need to run at various power levels during its operational life. The flame temperature inside the combustor is dependent on the power level and will increase or decrease depending on the power setting. To see the effect of flame temperature on combustion oscillations, another run was done with a parametric sweep of the flame temperature.

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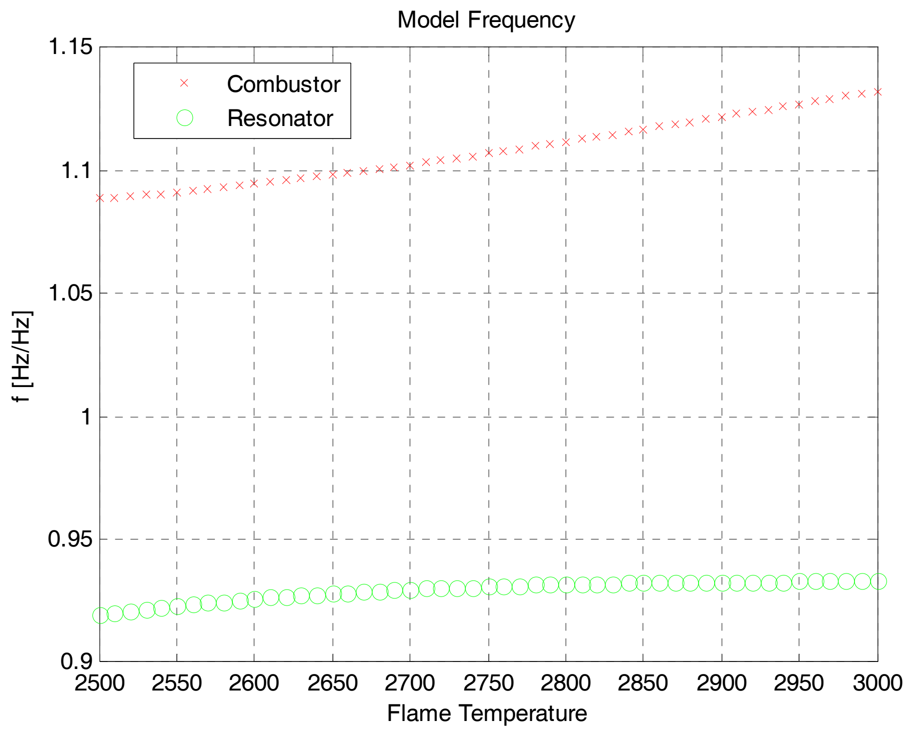

Figure 14: Frequency shift from flame temperature variation (predicted)

Figure 15 shows that the combustor's normalized frequency will vary from 1.09 to 1.13 during this sweep in flame temperature. Hence the resonators may only be effective in a certain temperature band, depending on their natural frequency.

Conclusions

A low order acoustic model was written in Matlab that incorporated the major components for the combustion chamber. Parametric studies were performed with the code in order to understand how the system's stability is affected by parameters like flame temperature, resonator volume, and resonator damping. It is recommended that further work on the heat release term be performed to enhance the model. It is hoped that the code and this report will be used as a guide in the final design any new combustor designs.